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| | | |
|---|---|---|
| 1. | Your reference | RJB/P470003GB |
| 2. | Patent application number *(The Patent Office will fill in this part)* | 0307038.0 27 MAR 2003 |
| 3. | Full name, address and postcode of the or of each applicant *(underline all surnames)* | **Torotrak (Development) Ltd.,** 1 Aston Way, Leyland, Lancashire, PR5 3UX. |
| | Patents ADP number *(if you know it)* | 737665002 |
| | If the applicant is a corporate body, give the country/state of its incorporation | ENGLAND. |
| 4. | Title of the invention | SYSTEM AND METHOD FOR CONTROLLING A CONTINUOUSLY VARIABLE TRANSMISSION |
| 5. | Name of your agent *(if you have one)* | **W.P.THOMPSON & CO.** |
| | "Address for service" in the United Kingdom to which all correspondence should be sent *(including the postcode)* | Coopers Building, Church Street, Liverpool, L1 3AB. |
| | Patents ADP number *(if you know it)* | 0000158001 |

| | | Country | Priority application number *(if you know it)* | Date of filing *(Day/month/year)* |
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| 6. | If you are declaring priority from one or more earlier patent applications, give the country and the date of filing of the or of each of these earlier applications and *(if you know it)* the or each application number. | | | |

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| 7. | If this application is divided or otherwise derived from an earlier UK application, give the number and the filing date of the earlier application | | |

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| Description | 27 |
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| Priority documents | |
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| Request for substantive examination *(Patents Form 10/77)* | |
| Any other documents *(Please specify)* | |

11. I/We request the grant of a patent on the basis of this application

Signature W.P. Thompson & Co. Date 26/03/2003

12. Name and daytime telephone number of person to contact in the United Kingdom R.J.BARTLE. 0151-709-3961.

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P47000390

DUPLICATE



# DESCRIPTION
# SYSTEM AND METHOD FOR CONTROLLING A CONTINUOUSLY VARIABLE TRANSMISSION

The present invention is concerned with the control of a transmission of continuously variable ratio, torque-controlled type and of an associated engine.

The term "engine" as used herein should be understood to encompass any suitable device for providing rotational drive including internal combustion engines and electric motors. The present invention has been developed in connection with transmissions for motor vehicles and is particularly well suited to this application. Nonetheless it is considered potentially applicable to transmissions for use in other contexts.

In any continuously variable transmission there is a unit, referred to herein as a "variator", which provides a continuously variable drive ratio. The variator is controlled by means of a primary control signal applied to it. The variator couples to other parts of the transmission - typically gearing leading on one side of the variator to the engine and on the other side to driven components such as the driven wheels of a motor vehicle - through rotary input and output members. The ratio of rotational speeds of the input and output members is the "variator drive ratio".

The concept of "torque control" is known in this art but will now be explained. It

is useful to distinguish torque control from the alternative of "ratio control".

A *ratio*-controlled variator receives a control signal representing a required variator drive ratio. The variator responds by adjusting its drive ratio to the required value. The adjustment typically involves detecting the position of a ratio-determining element of the variator (e.g. the separation of the sheaves in a belt-and-sheave variator, or the position of the rollers in a toroidal-race type variator) and adjusting the actual position of this element to a desired position (determined by the control signal) using a feedback loop. Thus in a ratio controlled variator, ratio corresponds directly to the control signal.

This is not the case in a *torque*-controlled variator. Instead a torque-controlled variator is constructed and arranged such as to exert upon its input and output members torques which, for a given variator drive ratio, correspond directly to the control signal. It is torque which is the control variable rather than drive ratio. Changes in speed of the variator input and output, and hence changes in variator drive ratio, result from the application of these torques, added to externally applied torques (e.g. from engine and wheels), to the inertias coupled to the variator input and output. The variator drive ratio is permitted to change correspondingly.

Torque control has to date principally been applied to toroidal-race, rolling-

traction type variators. In an arrangement described for example in Torotrak (Development) Ltd's European patent EP444086, variator rollers serve to transmit drive between co-axially mounted input and output discs. The variator rollers exert respective torques $T_{in}$ and $T_{out}$ upon the input and output discs. Correspondingly the rollers experience a "reaction torque" $T_{in} + T_{out}$ about the disc axis. This reaction torque is opposed by an equal and opposite torque applied to the rollers about the axis by a set of actuators. The geometry is such that movement of the rollers about the disc axis is accompanied by "precession" of the rollers - a change in the angles of the roller axes to the disc axis, effecting a corresponding change in variator drive ratio. By controlling the actuator torque, the reaction torque $T_{in} + T_{out}$ is directly controlled. The control signal in this type of variator corresponds directly to the reaction torque.

The actual torques exerted by the variator upon its input and output depend not only on the control signal but also upon the prevailing drive ratio, since although the sum $T_{in} + T_{out}$ is uniquely determined by the control signal, the ratio $T_{in} / T_{out}$ is equal to the reciprocal of the variator drive ratio, and so subject to change with the variator drive ratio. Nonetheless it can be appreciated that, for a given drive ratio, both $T_{in}$ and $T_{out}$ are uniquely determined by the control signal.

The direct correspondence between reaction torque and control signal is not provided by all torque-controlled variators. An example of a torque-controlled variator of a quite different type, using belt-and-sheave construction, is provided

in the applicant's own prior European patent 736153 and its US counterpart 5766105, wherein one sheave is mounted upon its drive shaft in such a way that motion of the sheave relative to the shaft along a helical path is permitted. Hence when torque is applied to the sheave, a corresponding force along the axis of the shaft is created. This axial force is opposed by a force applied to the sheave by an actuator. Again, an equilibrium is created between the two forces. It can again be said of this example that the torque $T_{in}$ exerted by the sheaves upon the shaft is, for a given variator drive ratio, uniquely determined by the control signal, which corresponds to the force applied by the actuator.

A feature common to both arrangements is that the variator comprises a component - the movable sheave or variator roller - whose position corresponds to the prevailing variator drive ratio and that this component is subject to a biasing torque (or force) which is determined by the control signal and is balanced by the torques created at the variator input/output.

Effective utilization of torque-controlled transmissions depends on electronics to regulate the engine and transmission in unison. Early papers on the electronic control of such a power train are by Stubbs - "The Development of a Perbury Traction Transmission for Motor Car Applications", ASME (The American Society of Mechanical Engineers) paper no. 80-GT-22, March 1980 and also by Ironside and Stubbs - "Microcomputer Control of an Automotive Perbury Transmission", IMechE paper no. C200/81, 1981. Both papers describe a project

concerned with electronic control of a transmission based on a toroidal-race rolling-traction type variator operating in torque controlled mode.

Both papers point out an important advantage associated with continuously variable transmissions: that fuel economy can be greatly enhanced when using such transmissions by operating the engine at or close to the levels of engine speed and engine torque at which it is most fuel efficient. For any given level of engine power demanded by the driver there is a particular combination of engine speed and engine torque which provides best fuel efficiency. Stubbs plotted the locus of such "optimal efficiency" points on a graph forming a line representing the optimal engine efficiency. The control strategies proposed by Ironside and Stubbs were based on operating the engine on this line where possible.

In the control schemes described in these papers the driver's demand was interpreted as a requirement for wheel torque, which was then converted into a requirement for engine power by multiplication by the rotational speed of the vehicle wheels. From this power a unique point on the optimal efficiency line was selected, providing target values for the engine torque and engine speed. The engine was set to produce the target torque and the loading applied to the engine by the variator adjusted to bring the engine speed to the target value, using a closed loop based on engine speed.

Testing showed this simple closed loop engine speed control to be inadequate.

Typically such a system is either too slow in its response or is unstable, leading to oscillation of engine speed about the target value.

The challenges involved in controlling a torque-controlled transmission are very different from those involved in controlling a ratio-controlled transmission. In the latter, since the variator maintains a chosen drive ratio, torque at the driven wheels is related directly to engine torque. Engine speed control is a relatively straightforward matter since, by maintaining a set drive ratio, the transmission provides a direct relationship between engine speed and vehicle speed. In a torque-controlled transmission, in which drive ratio is not the control variable and is permitted to vary, the engine and wheels can be thought of as being effectively de-coupled from one another. Wheel torque is controlled by the variator rather than by engine torque. Engine speed is not constrained to follow vehicle speed.

Instead the control signal applied to the variator determines a loading torque applied by the variator to the engine. Combustion within the engine creates an engine torque. The sum of the loading torque and the engine torque acts upon the inertia referred to the engine (contributed by masses in both engine and the transmission) and so determines engine acceleration. While the loading torque and the engine torque are equal, engine speed is constant. Changes in engine speed result from an inequality between these torques. Dynamic matching of engine torque to loading torque is thus fundamental to management of the drive line as a whole and of engine speed in particular. Failure to manage the balance

between engine torque and loading torque can lead to rapid and unintended changes in engine speed.

The profile of changes in engine speed is important to the "driveability" of the vehicle. The fact that in a CVT power train the engine is typically run at low speed and high torque (to provide high fuel economy) makes the management of engine speed especially important. When the driver calls for an increase in power the engine, already operating close to its maximum torque, must typically be accelerated in a controlled manner in order to be capable of providing the required power.

An object of the present invention is to make possible effective control of a drive line utilizing a torque-controlled transmission.

In accordance with a first aspect of the present invention there is a method of controlling a continuously variable ratio transmission of the type comprising a continuously variable ratio unit ("variator") which has rotary input and output members through which the variator is coupled between an engine and a driven component, the variator receiving a primary control signal and being constructed and arranged such as to exert upon its input and output members torques which, for a given variator drive ratio, correspond directly to the control signal, the method comprising:

determining a target engine acceleration,

determining settings of the variator's primary control signal and of an engine torque control for providing the required engine acceleration and adjusting the control signal and/or the engine torque control based on these settings,

predicting a consequent engine speed change, allowing for engine and/or transmission characteristics, and

correcting the settings of the control signal and engine torque based on a comparison of actual and predicted engine speeds.

In accordance with a second aspect of the present invention there is a method of controlling a continuously variable ratio transmission of the type comprising a continuously variable ratio unit ("variator") which has rotary input and output members through which the variator is coupled between an engine and a driven component, the variator receiving a primary control signal and being constructed and arranged such as to exert upon its input and output members torques which, for a given variator drive ratio, correspond directly to the control signal, the method comprising:

determining a target engine acceleration,

determining an excess torque TrqAcc required to accelerate power train inertia to achieve the target engine acceleration, and

adjusting the control signal to the variator and/or adjusting a torque controller of the engine such that engine torque is equal to loading torque applied by the transmission to the engine plus the excess torque TrqAcc.

Specific embodiments of the present invention will now be described by way of example only, with reference to the accompanying drawings, in which:-

Figure 1 is a highly simplified representation of a power train of torque-controlled type suitable for implementing the present invention;

Figures 2a and 2b are graphs representing the interpretation of a driver's input in a control system embodying the present invention;

Figure 3 is a schematic representation of part of a power train control system by which dynamic matching of engine torque and loading torque is carried out, in accordance with the present invention;

Figure 4 is a schematic illustration of closed loop engine speed control logic forming part of the same control system;

Figure 5 is a highly simplified representation of the hardware used in power train control;

Figure 6 is a simplified representation of a toroidal-race, rolling-traction variator, of a type which is in itself known, suitable for use in implementing the

present invention;

Figure 7 is a schematic representation of a multi-regime transmission operable in accordance with the present invention; and

Figure 8 is a map of engine torque versus engine speed for an internal combustion engine.

The present invention has been developed in connection with a vehicle transmission using a torque-controlled variator of toroidal-race, rolling-traction type. The invention is considered potentially applicable to other types of torque-controlled transmission. Nonetheless the toroidal-race variator in question will now be very briefly described, in order to illustrate certain relevant principles. More detail on both the construction and function of this type of variator is to be found in various patents and published applications held by Torotrak (Development) Ltd. including European patent EP444086.

Figure 6 illustrates some of the major components of the variator 10 and Figure 1 illustrates, in highly schematic format, major parts of a drive line incorporating the variator. In Figure 6 the variator is seen to comprise co-axially mounted input and output discs 12, 14 together defining a toroidal cavity 22 containing a variator roller 20. The roller runs on respective faces of the input and output discs in order to transmit drive from one to the other. The roller is mounted in a manner permitting it to move along a circumferential direction about the axis 24 of the discs 12, 14. The roller is also able to "precess". That is, the roller's axis is able

to rotate, changing the inclination of the roller to the disc axis. In the illustrated example the roller is mounted in a carriage 26 coupled by a stem 28 to a piston 30 of an actuator 32. A line from the centre of the piston 30 to the centre of the roller 20 constitutes a "precession axis" about which the whole assembly can rotate. Changes in the inclination of the roller result in changes in the radii of the paths traced on the input and output discs 12, 14 by the roller. Consequently a change in roller inclination is accompanied by a change in variator drive ratio.

Note that the precession axis does not lie precisely in a plane perpendicular to the disc axis, but is instead angled to this plane. This angle, labeled CA in Figure 6, is referred to herein as the "castor angle". The roller's mounting permits it to move the centre of the roller following a circular path centred on the disc axis. Furthermore, the action of the discs 12, 14 upon the rollers tends to maintain the rollers at such an inclination that the roller axis intersects the disc axis. This intersection of the axes can be maintained, despite movement of the roller along its circular path, by virtue of the castor angle. The result is that translational movement of the roller about the disc axis is accompanied by precession of the roller and so by a change in variator drive ratio. If one neglects slip between the roller and the discs, the position of the variator roller corresponds to the variator drive ratio and so to the speed ratio between the engine and the driven wheels.

The actuator 32 receives opposed hydraulic fluid pressures through lines 34, 36 and force applied to the roller by the actuator to the roller corresponds to the

difference in pressures in the lines. This pressure difference is the principal control signal applied to the variator, in this example. The effect of this force is to urge the roller to move along its circular path about the disc axis. Equivalently one can say that the actuator exerts a torque about the disc axis upon the roller. The actuator torque is balanced by torque created by the interaction of the roller with the discs. The roller exerts a torque $T_{in}$ upon the input discs 12 and a torque $T_{out}$ upon the output disc 14. Correspondingly the discs together exert a torque $T_{in} + T_{out}$ upon the roller, about the disc axis. The quantity $T_{in} + T_{out}$ (the reaction torque) is at all times equal to the actuator torque and so directly proportional to the control signal formed by the aforementioned pressure difference.

Looking now at Figure 1, an engine is represented by box 16 and is coupled to the input disc 12 of the variator. A direct coupling is shown in this highly simplified drawing. In practice there is of course intervening gearing and this aspect of the transmission will be considered below. Masses coupled to the variator's input disc, including the mass of the engine itself, provide an input inertia $J_i$ Masses coupled to the output disc 14 provide an output inertia $J_0$.

The control signal determines, at the prevailing variator drive ratio, the loading torque $T_{in}$ applied to the variator input disc 12 by the roller. In the simplified arrangement illustrated in Figure 1 the variator input disc 12 is seen to be directly coupled to the engine so that the loading torque applied to the engine is equal to the loading torque $T_{in}$ applied to the variator input disc 12. Of course in a

practical transmission gearing is interposed between the input disc 12 and the engine. Consequently the loading torque $T_{load}$ experienced by the engine is equal to the variator input torque $T_{in}$ divided by the ratio of the intervening gearing.

Under normal operating conditions the loading torque $T_{load}$ is opposed by the engine torque $T_e$. The sum of these two torques acts upon the driveline input inertia $J_v$ (which includes the engine inertia) so that an inequality between $T_{load}$ and $T_e$ causes a change in engine speed $\omega_e$. The variator automatically accommodates resultant change in transmission ratio. Likewise the control signal determines the variator output torque $T_{out}$. This is added to externally applied torques $T_v$ (e.g. from the vehicle wheels) in determining the net torque available to accelerate the output inertia $J_o$ (which, while the vehicle wheels maintain traction with the road, includes the vehicle's inertia). In this way changes in transmission output speed $\omega_v$ are produced.

The illustrated variator 10 is of course greatly simplified for the sake of clarity. For instance a practical variator typically has two pairs of input/output discs defining two toroidal cavities each of which contains a set of rollers. In such an arrangement the reaction torque is the sum of the torques applied to all of the variator rollers. The principles of operation set out above are however essentially unchanged in a practical transmission.

A broad overview of the main functional components of a control system

embodying the presented invention is provided in Figure 5 wherein the engine is seen at 500 and drives a continuously variable, torque-controlled type transmission 502. The diagram schematically indicates both the variator 504 and epicyclic gearing 506 through which the variator is coupled between transmission input and output in either of a low regime, in which the range of ratios available from the variator maps onto a low range of overall transmission ratios, and a high regime, in which the range of variator ratios maps onto a higher range of overall transmission ratios. The transmission output is coupled to a load - typically the driven wheels of a motor vehicle - which is represented by block 508 in the drawing.

The control of both engine and transmission is performed electronically, subject to direction from the driver. Conventional digital microprocessors are programmed for this task in current embodiments. The illustrated architecture serves as an example only and may be further simplified in production versions, but comprises an electronic Power train Control Unit ("PCU") which receives data from instrumentation associated with the engine, the transmission and also from the driver's control 509 (formed e.g. by the accelerator pedal of a conventional motor vehicle). In response the PCU provides outputs controlling the behaviour of both engine and transmission. Engine control is carried out through an electronic engine torque controller 510 and throttle 512. Transmission control is effected in this exemplary embodiment by control of hydraulic pressures applied to the variator 504 and, in order to control transmission regime, to clutches of its

associated gearing 506.

In controlling a motor vehicle power train it is necessary firstly to interpret the driver's input, which is of course typically communicated through the position of an accelerator pedal (and for the sake of brevity a pedal will be referred to below, although the driver control could in practice take some other form). What the current control system does is to map pedal position onto a driver demand for wheel torque and engine speed, taking account of vehicle speed. Figure 2a is a graph showing the engine speed requested by the driver (SpdEngDr) against the prevailing vehicle speed (SpdVeh) and the pedal positions (PosPedal). Figure 2b shows the driver's requested wheel torque (TrqWheelDr) once more against vehicle speed and pedal position. The two graphs are recorded in the form of look up tables in the control system.

Based on the driver's requested wheel torque - TrqWheelDr - a mathematical model of the transmission (taking account of factors including transmission efficiency) is used to obtain the driver's requested engine torque which, in conjunction with the driver's requested engine speed, enables the driver's requested engine power to be determined. The driver's requested engine torque and engine speed may be used unmodified or alternatively the driver's requested engine power may be used in conjunction with an engine map, or set of engine maps, to determine the optimal engine speed and engine torque for providing the requested engine power. Purely by way of example, to illustrate how optimization

with regard to engine efficiency can be achieved, Figure 8 is an engine map with engine speeds along the horizontal axis and engine torque T on the vertical axis. Line 800 shows how the driver's requested engine speed and torque vary as the pedal position is changed. Line 802 represents the relationship between engine speed and engine torque which provides optimal engine efficiency. Asterisks on the two lines correspond to the same level of engine power, and the system can choose between the two operating points.

The process of interpretation of driver demand results in a target engine torque, TrqEngBaseReq and a target engine speed SpdEngBaseReq.

In "steady state" operating conditions the engine is operated by the system at the target levels of engine torque and engine speed. However vehicle operating conditions- vehicle speed, driver demand etc. - are in practice rapidly changing.

Under "transient" conditions, where the engine departs from the target torque and speed, the task of the system under consideration is to control the engine and transmission in such a manner as to achieve, or in a dynamic situation at least to adjust toward, these values. The control process is illustrated schematically in Fig. 3 but can be summarised as comprising the following steps, which are repeated in a loop.

1. Determine the difference between actual and target engine speeds.

2. Calculate from this difference a target engine acceleration - i.e. the rate at which the engine should be accelerated toward the target engine speed (a controlled engine speed profile is desired) and then calculate the torque which will be taken up in overcoming inertia in order to provide the target engine acceleration (based on the moment of inertia ,of the engine and transmission, referred to the engine).

3. Set the engine torque controller appropriately to provide the engine torque required both to (1) create an appropriate wheel torque and (2) accelerate the engine. Where possible the wheel torque thereby created corresponds to the driver request. However, the available engine torque being finite, it is necessary in some situations to accept a lower wheel torque in order to provide the torque required to accelerate the engine.

4. Calculate what instantaneous torque the engine will actually provide given this engine torque controller setting, since the engine's reaction to its controller is not instantaneous. Factors including the engine's intake manifold dynamics create a lag between engine adjustment and engine torque. Techniques for modeling the instantaneous output torque are known in the art and are applied here.

5. Adjust the control signal applied to the variator to load the engine with a torque equal to the calculated instantaneous engine torque, derived from the aforesaid model, minus the torque required to accelerate the engine, calculated at step 2.

6. Calculate what engine acceleration is actually expected. This expected value does not precisely match the target acceleration, since the calculation of the expected value takes account of (a) the instantaneous engine torque calculated above and (b) a further model representing the transmission's response to the control applied at step 5 above, the transmission too having a time lag in its response to the control input. The calculation is also based on the moment of inertia of the engine and transmission referred to the engine.

7. Integrate the engine acceleration obtained at step 6 to obtain a predicted engine speed, and then apply a closed loop correction to the engine speed, correcting it toward the predicted value.

Because the closed loop engine speed correction is used only to adjust the engine speed toward an expected value based on models of the engine and transmission dynamics, the amount of such correction is minimized. The process allows the engine acceleration to be controlled and "profiled" (the rate of engine acceleration being a function of the discrepancy between actual and target engine speeds) in a highly effective manner.

The moment of inertia referred to the engine is not constant but instead is a function of transmission ratio and also of the rate of change of ratio and the prevailing transmission regime. Calculation of the torque required to accelerate the engine will be described below.

The control process will now be described in more detail with reference to Fig. 3, wherein the target engine torque is represented by the input variable TrqEngBaseReq and the target engine speed by the input variable SpdEngBaseReq.

Looking firstly at the top left of the diagram, TrqEngBaseReq is added, at 200, to a torque TrqAcc calculated to provide a target engine acceleration. The determination of TrqAcc will be considered below. Of course the torque available from the engine is finite and a limiter 202 ensures that if the input to the limiter is a torque greater, or indeed smaller, than the engine can supply then it is modified to fall within the available torque range. The resulting required torque value, TrqEngReq, is supplied to the engine torque control CmdPosThrottle of the engine controller. Hence, where possible, the engine is set to provide an engine torque corresponding to the sum of the target engine torque, TrqEngBaseReq, and the torque TrqAcc necessary to accelerate the engine toward the target engine speed.

The engine's response to the engine torque controller is not instantaneous. Even neglecting the effects of engine inertia, the torque generated by the engine lags somewhat being the throttle input, as is well known to the skilled person. Such a time lag is potentially problematic in a torque-controlled transmission, where even a brief mismatch between engine torque and variator reaction torque (and correspondingly in the loading torque applied by the transmission to the engine)

can lead to a dramatic error in the engine speed, as explained above. To avoid such problems the illustrated control system incorporates an engine model 204 which, based on the torque requirement TrqEngReq input to the engine controller and on a model of the engine behaviour, outputs an estimate TrqEngEst of the instantaneous torque created by the engine, allowing for the time lag in the engine's response to its torque controller.

At 206 the torque TrqAcc necessary to accelerate the inertia of the engine and transmission is subtracted from the instantaneous engine torque TrqEngEst to give the loading torque to be applied to the engine by the transmission, from which the reaction torque required of the variator is then obtained. However the reaction torque is modified by means of a latching strategy 208 in order to prevent unwanted variations in wheel torque under certain conditions.

The latching strategy serves in particular to sustain wheel torque during brief delays in the engine's response to its torque controller. Consider for example the situation where the transmission has been operating in "over-run" (providing engine braking - ie. negative wheel torque) and the driver then depresses the pedal to demand positive wheel torque. The control system responds with a demand for increased engine torque but this is not available instantaneously and to provide the driver with a rapid response to his pedal input the required variator reaction torque is modified by the latching strategy.

Hence the latching strategy serves to limit deviation of wheel torque from the level demanded by the driver.

The output from the latching strategy represents the engine loading torque to be provided by virtue of the variator and this is converted at 210 to a pressure differential for application to the variator (the variator's primary control signal) which is passed, as an output variable TrqReacVarReq, to logic controlling the fluid pressures applied to the variator itself.

The control system as so far described provides outputs to control both the engine torque controller and the transmission hydraulics. Based on these two outputs the consequent changes in engine speed are estimated. In doing so it is necessary to take account not only of the time lag in the engine's response to its torque controller, modeled at 204 as mentioned above, but also of time lags in the response of the variator to its control input. As already explained the control signal to the variator is provided in the form of two oil pressures controlled by valves in the hydraulics associated with the variator. Changes in the valve settings take a finite time to produce an effect, this delay being allowed for at 212. Compliance in the hydraulics also produces a contribution to the lag, modeled at 214 to produce an output which is an estimate of the instantaneous transmission reaction torque.

The torque available to overcome power train inertia and so accelerate the engine

is the difference between the instantaneous loading torque applied to the engine (which could equivalently be referred to as the torque input to the transmission) and the instantaneous engine torque. In Figure 3 a comparator 216 takes away the estimated instantaneous engine torque, output from the engine model 204, from the estimated instantaneous loading torque. Dividing the result by the inertia J referred to the engine at 218 gives an estimate of engine acceleration and integrating at 221 provides a prediction of the engine speed. In practice, since the inertia J is not constant, this calculation is somewhat more complicated, as will be explained below. The integrator also receives the target engine speed SpdEngBaseReq, which serves to saturate the integrator, capping the engine speed at the desired value.

This prediction is output as the variable SpdEngReq to closed loops controlling the engine and the variator which effect adjustments to the engine torque controller setting and to the variator reaction torque to adjust the engine speed toward the predicted value. The effect is to use the closed loop only to minimize the deviation of the engine's actual behaviour from the behaviour which it is predicted to exhibit, based on the torque controller and variator settings. This is highly advantageous in minimizing the correction which the closed loop control is required to make and results in stable power train control.

It has yet to be explained how the target engine acceleration is determined. Note that the target engine speed SpdEngBaseReq is supplied, via a limiter 219, to

subtraction block 220 which takes the predicted engine speed SpdEngReq away from the limited target engine speed SpdEngBaseReqLimit, giving a prediction of the difference between the actual engine speed and the target engine speed. The system controls the engine acceleration as a function of this difference. In the illustrated example, the target engine acceleration is chosen to be proportional to the difference SpdEngBaseReqLimit - SpdEngReq, a constant of proportionality GainAccEng being introduced at 222. This process provides a suitable profile to the engine acceleration, which is large when the engine's speed is a long way from the target value and falls as the engine speed approaches the target value. Clearly, however, a different function could be chosen for setting the target engine acceleration AccEng.

A further limiter 224 ensures that the desired engine acceleration does not exceed acceptable limits. It is then necessary to calculate TrqAcc, the excess torque required to achieve the engine acceleration AccEng. Of course TrqAcc is equal to AccEng multiplied by the driveline inertia J referred to the engine. However J is in a practical transmission not constant. An explanation will now be provided of how the relationship between TrqAcc and engine acceleration can be calculated.

This relationship arises from the particular form of the gearing used to couple the variator to the engine and wheels and Figure 7 provides a schematic illustration of a suitable arrangement. This is of the dual regime, power recirculatory type already known in the art e.g. from Torotrak (Development) Limited's earlier

patents including EP933284. In Figure 7 the engine is indicated at 700, the variator at 702 and an output of the transmission to driven vehicle wheels at 704. An epicyclic "shunt" gearing arrangement is indicated at 706 and boxes $R_1$-$R_4$ represent gear ratios at various points in the transmission.

The epicyclic comprises, in the usual manner, a planet carrier CAR, a sun gear SUN and an annular outer gear ANN. The planet carrier CAR is driven from the engine via gearing $R_1$, $R_3$. The sun gear is driven via $R_1$, $R_2$ and the variator 702 itself. The prevailing variator ratio will be referred to as $R_v$.

To engage low regime (in which the available range of variator drive ratio maps onto a low range of transmission ratios) a low regime clutch LC is engaged, coupling the annular gear ANN to the output 704 via gearing with a ratio $R_4$. In low regime power is recirculated through the variator in a manner familiar to the skilled person.

To engage high regime (in which the available range of variator drive ratio maps onto a higher range of transmission ratios) a high regime clutch HC is engaged, forming a drive path from the variator output through clutch HC to gearing $R_4$ and so to the transmission output.

Inertias of the engine and transmission are represented by $J_1$, which includes the inertia of the engine; $J_2$, an inertia coupled to the sun gear SUN; and $J_3$, an inertia

coupled to the annular gear ANN. Rotational speeds of the three inertias are referred to respectively as $\omega_1$, $\omega_2$ and $\omega_3$. $\omega_1$ is therefore engine speed.

The relationship between TrqAcc and engine acceleration ($d\omega_1/dt$) is obtained using conservation of energy. An input power $\omega_1$ x TrqAcc goes to change kinetic energy of the transmission and changes in speed result.

Looking firstly at the low regime case, inertia $J_3$ is coupled to the vehicle wheels and is subject to the transmission output torque, which has of course been treated separately from TrqAcc. Hence it is necessary only to consider kinetic energies $Q_1$ and $Q_2$ of $J_1$ and $J_2$.

$$Q_1 = \tfrac{1}{2} J_1 \omega_1^2 \quad \text{and} \quad Q_2 = \tfrac{1}{2} J_2 \omega_2^2$$

and total kinetic energy

$$Q_{TOT} = \tfrac{1}{2} (J_1 \omega_1^2 + J_2 \omega_2^2) \text{ (Eq 1)}$$

and since the control system monitors variator ratio $R_v$, $\omega_2$ can be stated in terms of $\omega_1$

$$\omega_2 = R_1 R_2 R_v \omega_1 \qquad \text{(Eq 2)}$$

Substituting equation 1 into equation 2:

$$Q_{TOT} = (J_1 + J_2 (R_1 R_2 R_v)^2)\omega_1^2$$

and the rate of change of this kinetic energy is equal to the input power so:-

$$dQ_{TOT}/dt = TrqAcc \times \omega_1 \qquad = (J_1 + J_2 (R_1 R_2 R_v)^2)\omega_1 d\omega_1/dt$$
$$+ (2 J_2 R_1^2 R_2^2 R_v dR_v/dt)\omega_1^2/2$$

Hence it is possible to determine the excess torque TrqAcc required to accelerate the engine, this value being added to the target engine torque TrqEngBaseReq at 200, as already explained above.

In Figure 4 the logic implemented in the engine speed controller is schematically represented. In Figure 3 the calculation of three variables has been illustrated - (1) TrqEngReq, the required engine torque; (2) SpdEngReq, the required engine speed and (3) TrqReacVarReq, the required reaction torque. These three variables are received by the engine speed controller logic which controls both the engine and the transmission accordingly. Deviations of measured engine speed from the required value SpdEngReq are detected and corrected for in a closed loop control 300 which outputs corrections to both the engine torque and the variator reaction

pressure. Thus at 302 a correction from the closed loop control is added to the required engine torque TrqEngReq, the result of this addition being output to the engine torque controller 303 which in turn determines the "throttle position" and engine spark timing. At 304 a further correction from the closed loop control is added to the required reaction torque TrqReacVarReq and the result of this addition is output to a pressure controller 306, which itself operates in a closed loop based on comparison of measured and actual pressures in the hydraulic circuitry controlling the variator, adjusting valves in the circuitry to provide the required reaction pressures.

## CLAIMS

1. A method of controlling a continuously variable ratio transmission of the type comprising a continuously variable ratio unit ("variator") which has rotary input and output members though which the variator is coupled between an engine and a driven component, the variator receiving a primary control signal and being constructed and arranged such as to exert upon its input and output members torques which, for a given variator drive ratio, correspond directly to the control signal, the method comprising:

determining a target engine acceleration,

determining settings of the variator's primary control signal and of an engine torque control for providing the required engine acceleration and adjusting the control signal and/or the engine torque control based on these settings,

predicting a consequent engine speed change, allowing for engine and/or transmission characteristics, and

correcting the settings of the control signal and engine torque based on a comparison of actual and predicted engine speeds.

2. A method as claimed in claim 1 wherein the construction and arrangement of the variator is such that torques exerted by the variator upon its input and output members are proportional to magnitude of the primary control signal, for a given variator drive ratio.

3. A method as claimed in claim 1 wherein the construction and arrangement of the variator is such that the sum of the torques exerted by the variator upon its rotary input and output members is always proportional to magnitude of the primary control signal.

4. A method as claimed in any of claims 1 to 3 wherein the control signal takes the form of a difference between two hydraulic pressures.

5. A method as claimed in any of claims 1 to 4 wherein the target engine acceleration is calculated based on a difference between current and target engine speeds.

6. A method as claimed in any of claims 1 to 5 wherein target engine speed is set in dependence upon a user input.

7. A method as claimed in claim 6 wherein the user is interpreted as a demand for a transmission output torque and engine speed.

8. A method as claimed in claim 7 wherein the driver's demands for transmission output torque and engine speed are modified based on engine efficiency considerations.

9. A method as claimed in any of claims 1 to 8 wherein the demanded transmission output torque is converted to a target engine torque using a model of the transmission characteristics.

10. A method as claimed in any of claims 1 to 9 wherein, subject to limitations of the engine, a torque request to the engine torque controller is set to the sum of the target engine torque and the excess torque TrqAcc required to accelerate power train inertia.

11. A method as claimed in any of claims 1 to 10 wherein the engine's response to the torque controller is modelled to provide an estimate of instantaneous engine torque.

12. A method as claimed in claim 11 wherein the excess torque TrqAcc required to accelerate the engine is subtracted from the estimated instantaneous engine torque to obtain a required loading torque to be applied by the transmission to the engine, the variator control signal being adjusted to provide the required loading torque.

13. A method as claimed in any preceding claim wherein instantaneous values of engine torque and of loading torque applied to the engine by the transmission are estimated and used to calculate engine acceleration, the engine acceleration being integrated with respect to time to provide a prediction of engine speed, and closed loop control being applied to engine speed to correct it toward the predicted value.

14. A method of controlling a continuously variable ratio transmission of the type comprising a continuously variable ratio unit ("variator") which has rotary input and output members through which the variator is coupled between an engine and a driven component, the variator receiving a primary control signal and being constructed and arranged such as to exert upon its input and output members torques which, for a given variator drive ratio, correspond directly to the control signal, the method comprising:

determining a target engine acceleration,

determining an excess torque TrqAcc required to accelerate power train inertia to achieve the target engine acceleration, and

adjusting the control signal to the variator and/or adjusting a torque controller of the engine such that engine torque is equal to loading torque applied by the transmission to the engine plus the excess torque TrqAcc.

) C



15. A method as claimed in claim 14 wherein the construction and arrangement of the variator is such that torques exerted by the variator upon its input and output members is always proportional to magnitude of the primary control signal, for a given variator drive ratio.

16. A method as claimed in claim 14 wherein the construction and arrangement of the variator is such that the sum of the torques exerted by the variator upon its rotary input and output members is always proportional to magnitude of the primary signal control.

17. A method as claimed in any of claims 14 to 16 wherein the control signal takes the form of a difference between two hydraulic pressures.

18. A method as claimed in any of claims 14 to 17 wherein the target engine acceleration is calculated based on a difference between current and target engine speeds.

19. A method as claimed in any of claims 14 to 18 wherein target engine speed is set in dependence upon a user input.

25. A method as claimed in any of claims 14 to 24 wherein instantaneous values of engine torque and of loading torque applied to the engine by the transmission are estimated using engine and transmission models and used to calculate engine acceleration, the engine acceleration being integrated with respect to time to provide a prediction of engine speed and closed loop control being applied to the engine speed to correct it toward the predicted value.

## ABSTRACT

## SYSTEM AND METHOD FOR CONTROLLING A CONTINUOUSLY VARIABLE TRANSMISSION

There is described a method of controlling a continuously variable ratio transmission of the type comprising a continuously variable ratio unit ("variator") which has rotary input and output members though which the variator is coupled between an engine and a driven component, the variator receiving a primary control signal and being constructed and arranged such as to exert upon its input and output members torques which, for a given variator drive ratio, correspond directly to the control signal, the method comprising:

determining a target engine acceleration,

determining settings of the variator's primary control signal and of an engine torque control for providing the required engine acceleration and adjusting the control signal and/or the engine torque control based on these settings,

predicting a consequent engine speed change, allowing for engine and/or transmission characteristics, and

correcting the settings of the control signal and engine torque based on a comparison of actual and predicted engine speeds.

[FIG.3]

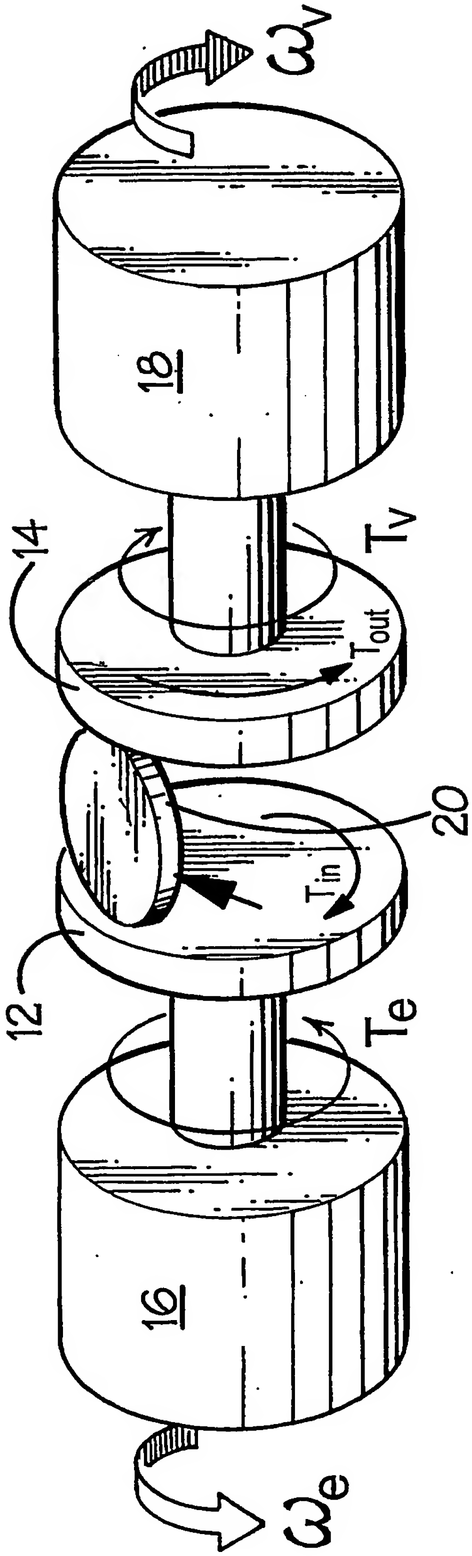


FIG.1.

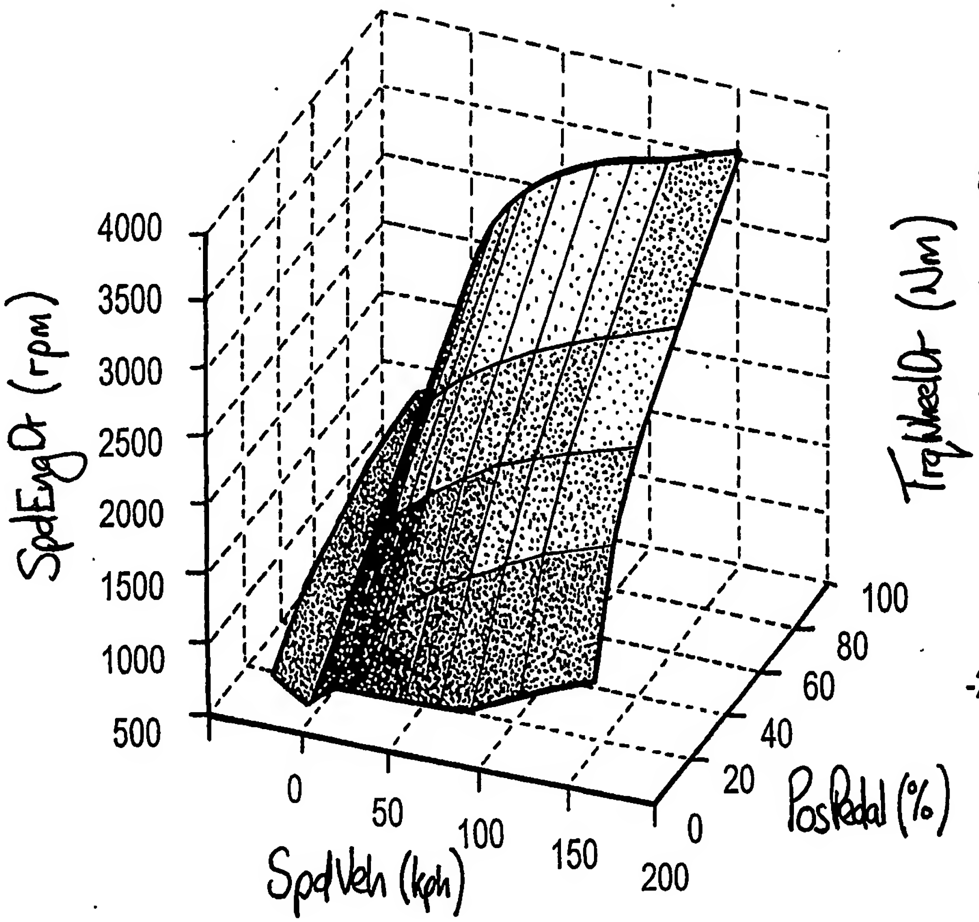
SpdEngDr (rpm)
4000
3500
3000
2500
2000
1500
1000
500
0
50
100
150
200
SpdVeh (kph)
0
20
40
60
80
100
PosPedal (%)

FIG.2a.

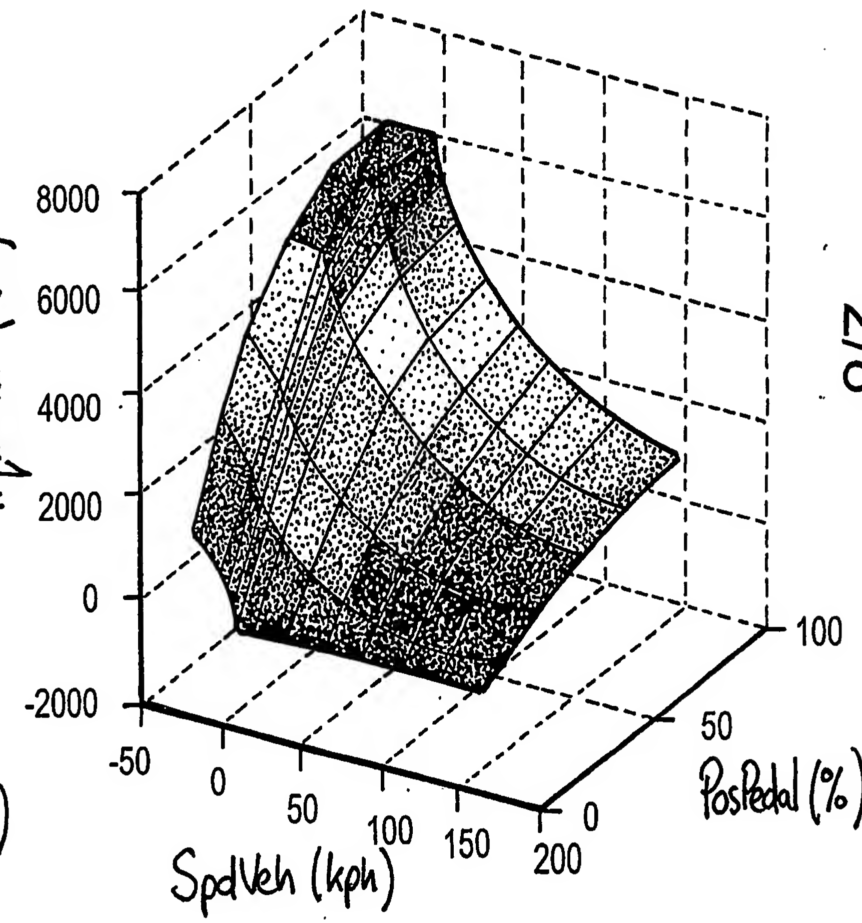
TrqWheelDr (Nm)
8000
6000
4000
2000
0
-2000
-50
0
50
100
150
200
SpdVeh (kph)
0
50
100
PosPedal (%)

FIG.2b.

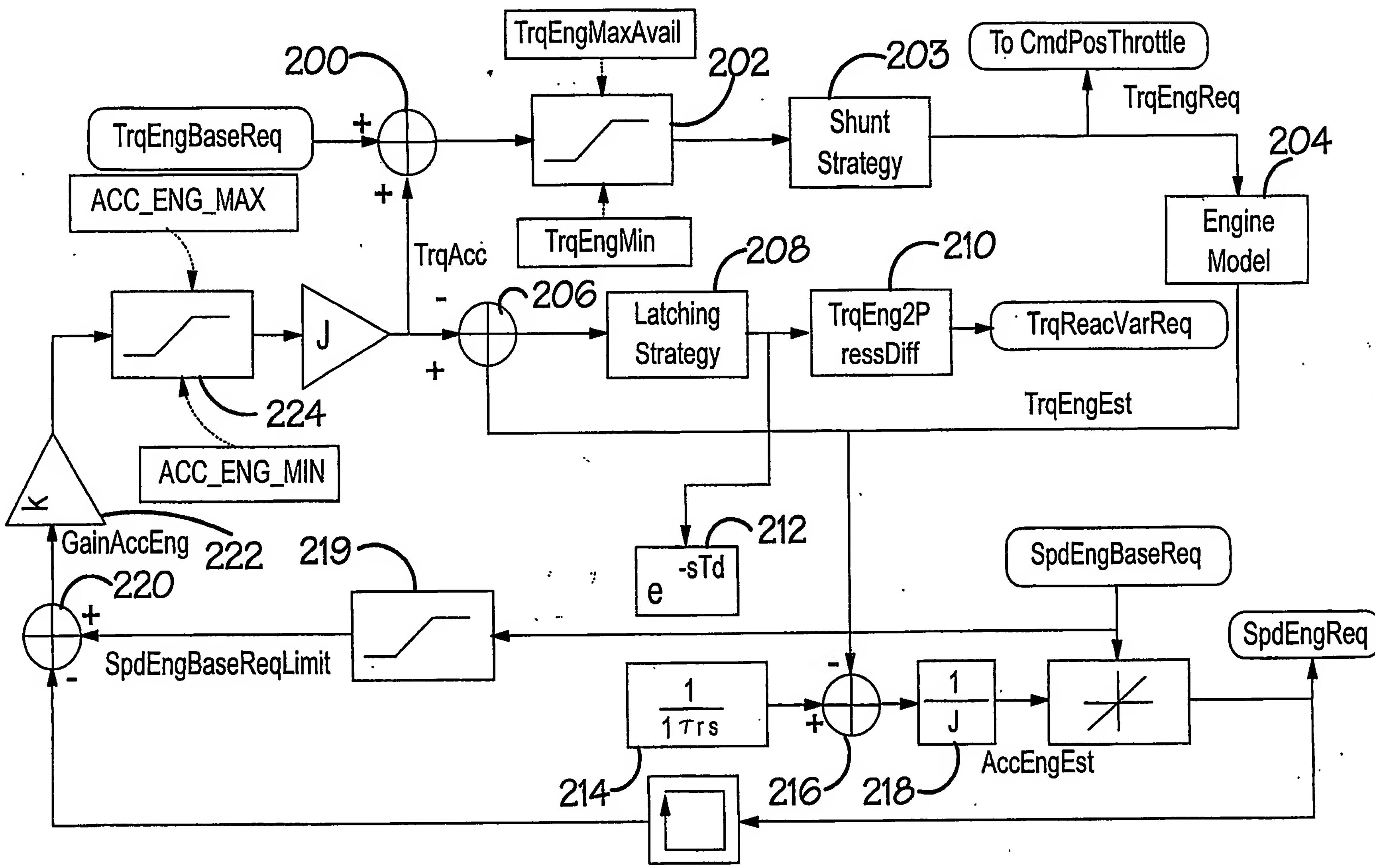
TrqEngBaseReq
ACC_ENG_MAX
ACC_ENG_MIN
TrqEngMaxAvail
TrqEngMin
TrqAcc
J
k
GainAccEng
Shunt Strategy
To CmdPosThrottle
TrqEngReq
Engine Model
Latching Strategy
TrqEng2PressDiff
TrqReacVarReq
TrqEngEst
e -sTd
SpdEngBaseReq
SpdEngReq
SpdEngBaseReqLimit
1
1τrs
1
J
AccEngEst
200
202
203
204
206
208
210
212
214
216
218
219
220
222
224
+
-

FIG.3.

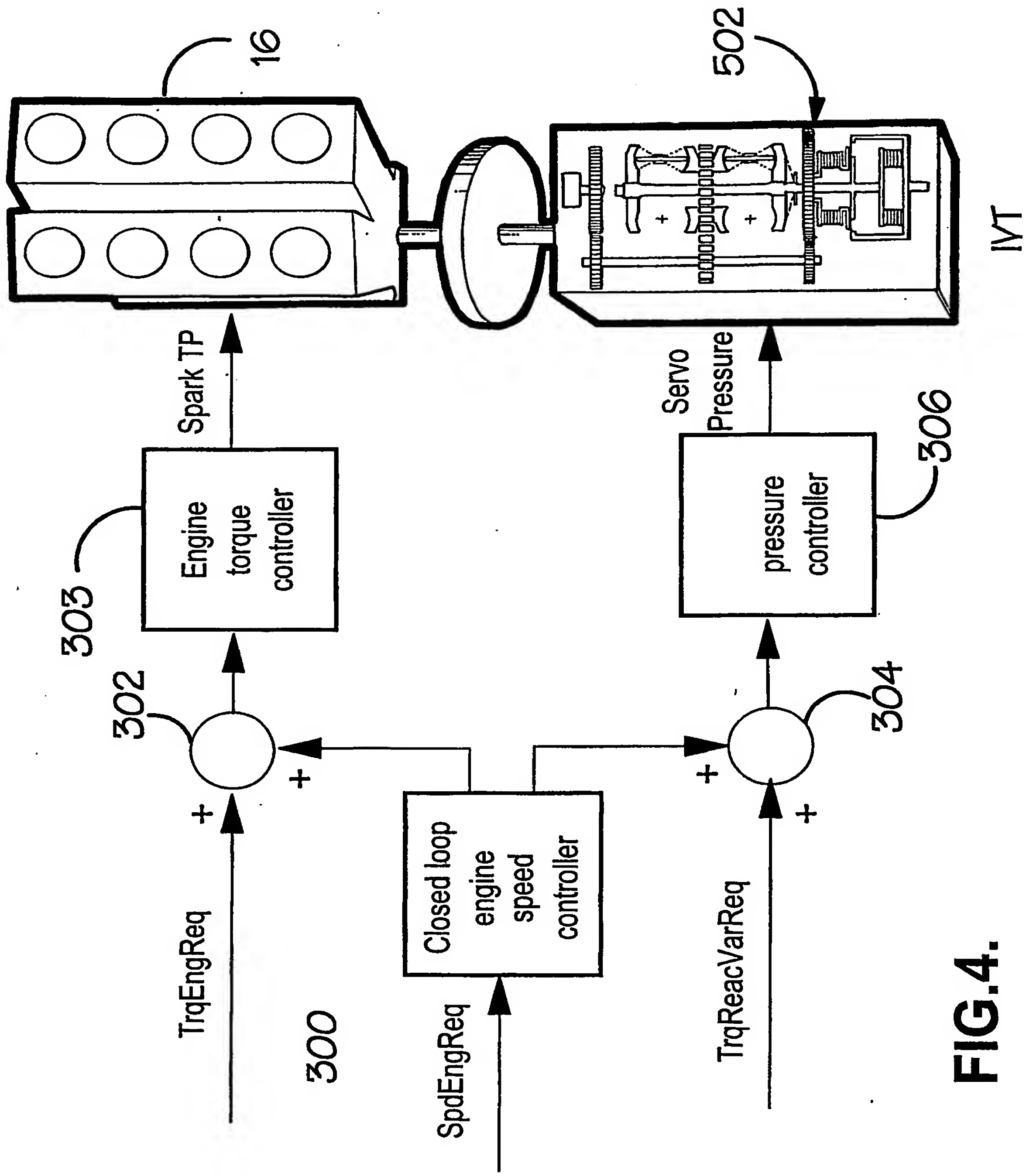
16
502
Spark TP
Servo
Pressure
IVT
Engine
torque
controller
303
pressure
controller
306
302
304
+
+
+
+
Closed loop
engine
speed
controller
TrqEngReq
300
SpdEngReq
TrqReacVarReq

FIG.4.

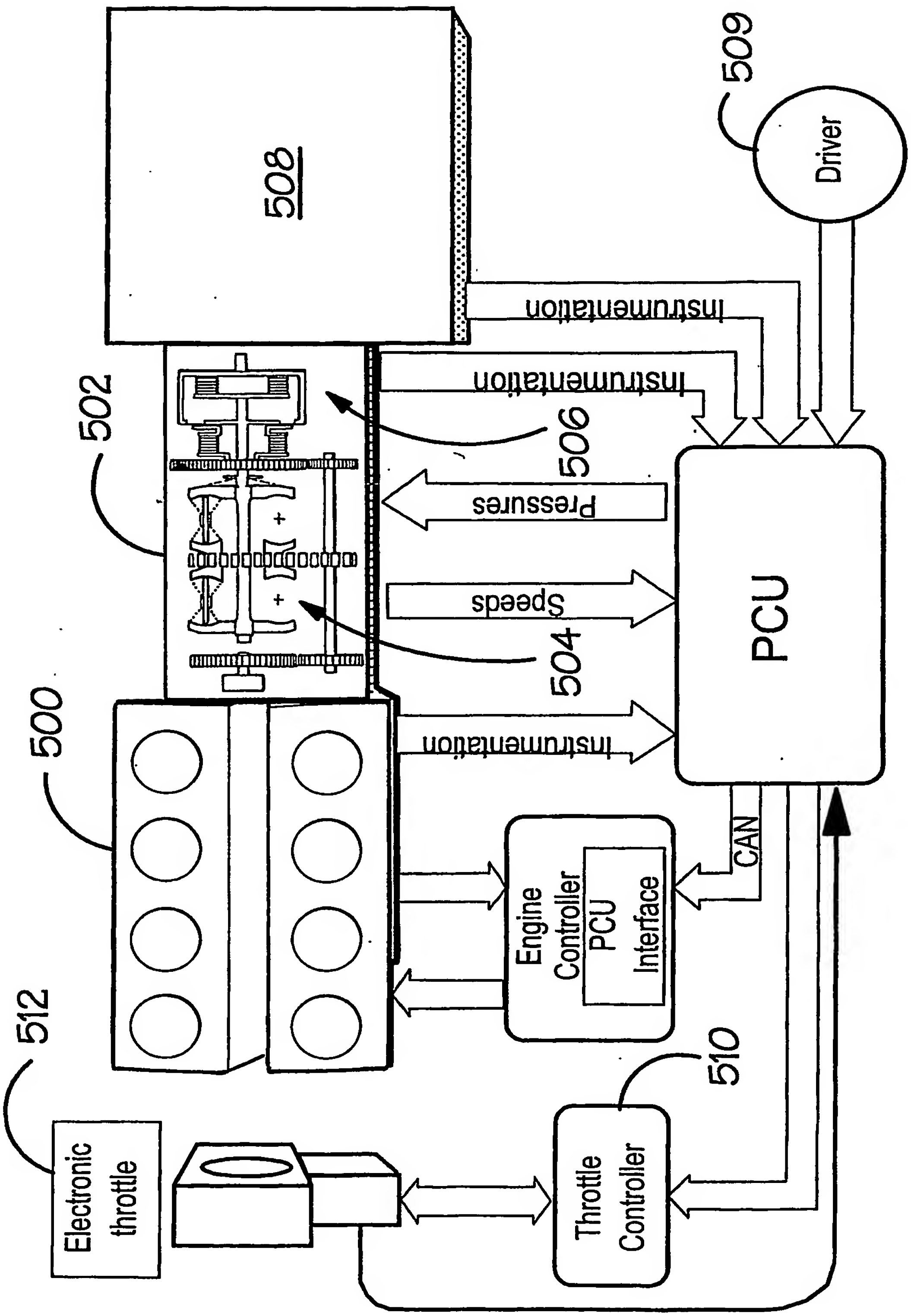
508
502
506
504
500
512
509
510
Driver
Instrumentation
Instrumentation
Pressures
Speeds
Instrumentation
PCU
CAN
Engine Controller
PCU Interface
Electronic throttle
Throttle Controller

FIG.5.

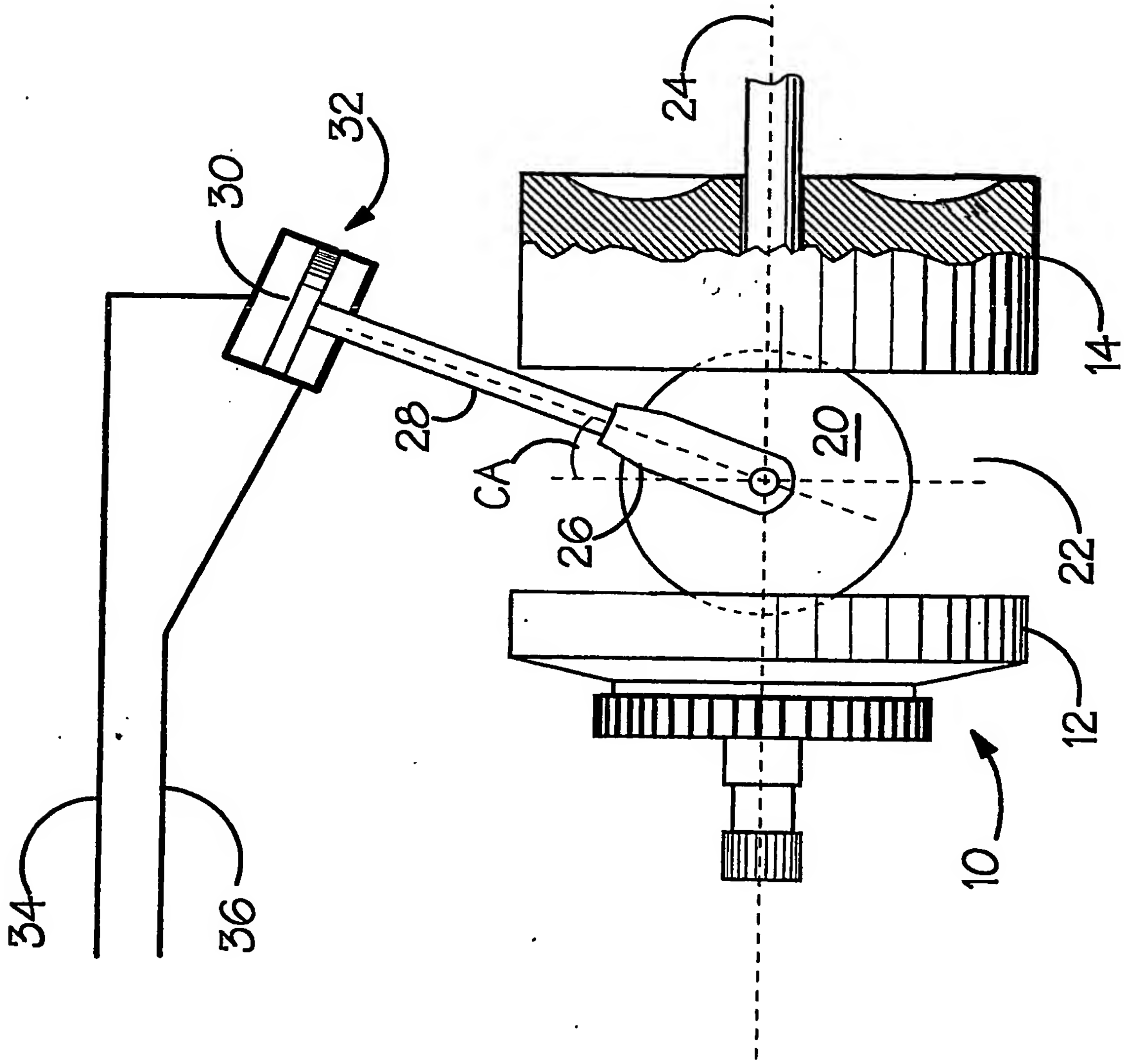


FIG.6.

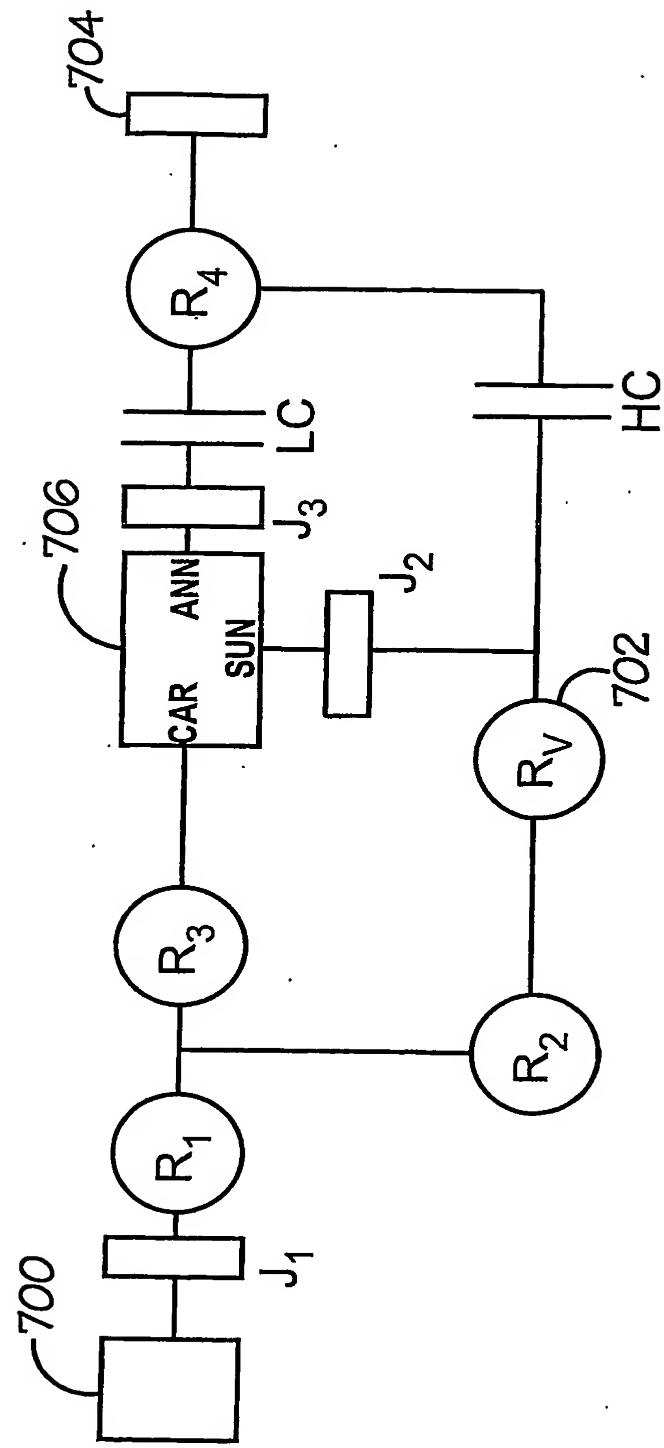


FIG.7.

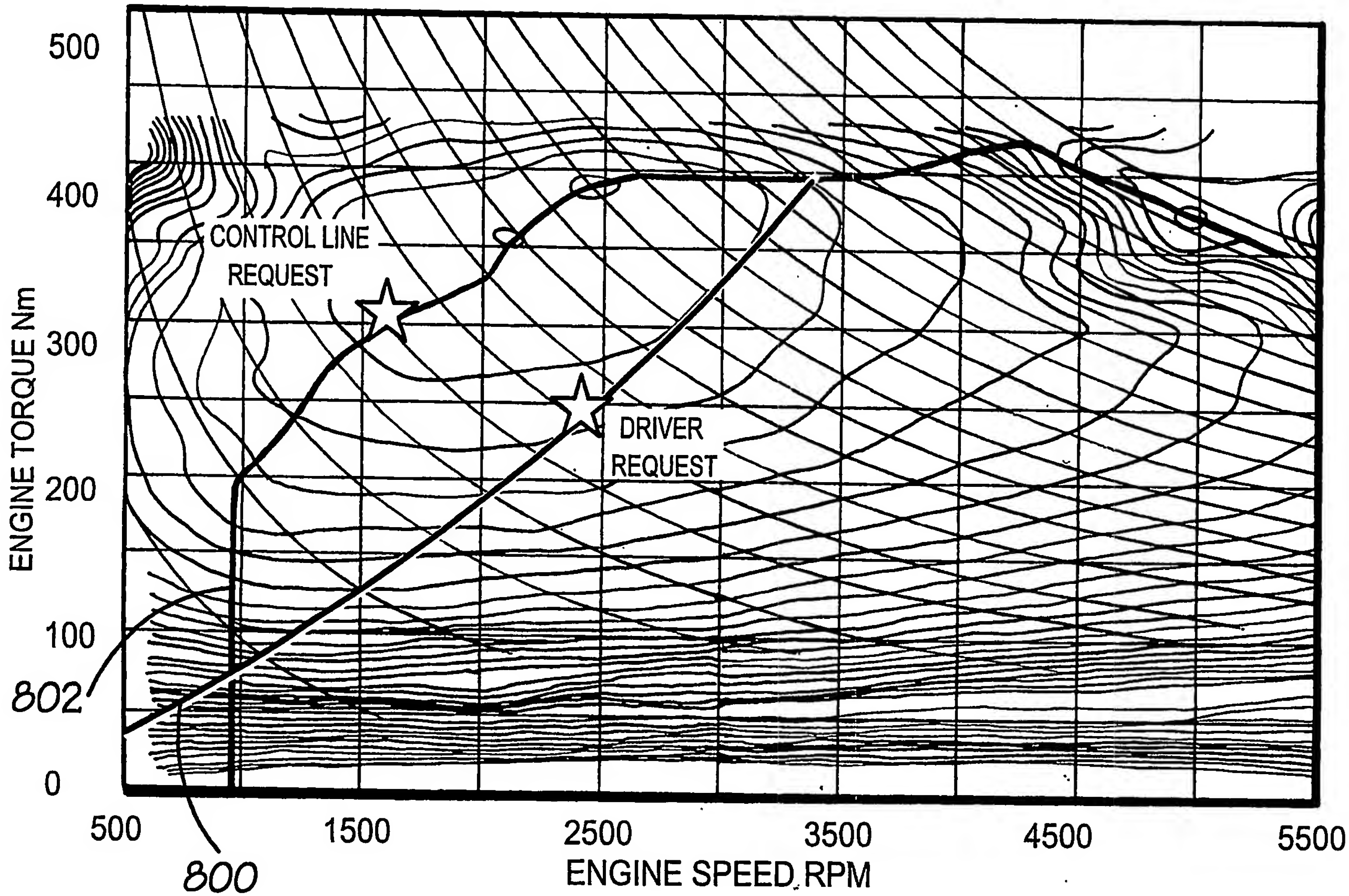
500
400
300
200
100
0
ENGINE TORQUE Nm
CONTROL LINE
REQUEST
DRIVER
REQUEST
802
800
500
1500
2500
3500
4500
5500
ENGINE SPEED RPM

FIG.8.